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A case study on thrust bearing failures at the SÃO SIMÃO hydroelectric power plant

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ABSTRACT

After twenty years without any apparent problems on their combined guide and thrust bearings, the six 280 MW hydrogenerators of the São Simão Hydroelectric Power Plant were failing. The source of the failure was the melting of the thrust pad babbitt lining. The machines began showing performance failures, leading to a sudden interruption in their operation. This caused considerable losses with high direct and indirect costs. The solution proposed by the bearing manufacturer was an improvement in the bearing design and the installation of new water–oil heat exchangers. The direct cost of their solution was estimated to be US $2,400,000.00. In a search for a less expensive alternative, CEMIG started a parallel study focused on the heat exchangers. A methodology based on heat transfer was applied, indicating that an increase in the heat exchange surface area could solve the problem. A third heat exchanger was added in one machine that already possessed two. The results fulfilled the preliminary predictions, eliminating the risk of additional babbitt lining failures. As a consequence of this success modeling, heat exchangers were replaced by stainless steel plate ones in all machines. This alternative solution had a total direct cost of US $600,000.00.

1. Introduction

The problem first appeared in the Unit 1 of the São Simão Hydroelectric Power Plant was due to melting of the babbitt pad linings and led to an emergency interruption on the machine operation, which was kept out of service until the pads were repaired. Failures were associated with high temperatures in the pads, up to 84 °C on summer time, when the temperature of cooling water can reach 30.8 °C. A similar problem had occurred in the other five machines of the plant [1]. CEMIG contacted the bearing manufacturer to evaluate and propose solutions for the issue. After lengthy testings, they recommended an improvement on bearing design and replacement of water–oil heat exchangers for a greater exchange capacity ones. But the high cost involved an average of US $400,000.00 (four hundred thousand dollars) per machine, led CEMIG to search for a less expensive alternative.

In an attempt to avoid new failures and subsequent interruptions in the machines' operation, the company was forced to limit its generators output in the summer until the problem was solved.

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Nomenclature

A	surface area, m^2
c	specific heat capacity, kJ °C^{-1} kg^{-1}
$LMTD$	log mean temperature difference, °C
p	pressure, kPa
$\dot{Q}$	heat transfer, kW
ρ	density, kg m^{-3}
$\overline{T}$	average temperature, °C
T	temperature, °C
U	overall coefficient of heat transfer, kW °C^{-1} m^{-2}
U^*	Modified overall coefficient of heat transfer (UA), kW °C^{-1}
$\dot{V}$	flux, m^3 s^{-1}
ν	viscosity, m^2 s^{-1}
$\dot{W}$	power, kW

Subscripts

1,2,3,4 and 5	see Figs. 2 and 3
b	bearing
e	heat exchanger
o	oil
p	pump
s	thrust pads
w	water

1.1. A new point of view

While searching for a simpler and cheaper alternative for the problem at hand and one that could be implemented in the shortest amount of time, the research targeted the water–oil heat exchangers as the best solution. An increase in the exchange surface area would decrease the temperature of the oil flowing out of the heat exchangers, providing the necessary decrease in the thrust pad temperature. Making use of an existent spare heat exchanger, it was considered adding it to Unit 1, with the oil circuit in series and water in parallel. Fig. 1 shows the original configuration of the oil cooling system, in which a third heat exchanger was placed.

1.2. Model

To determine the consequences of installing a third heat exchanger, a semi-empirical model was developed, based on values measured in Unit no. 1 with the original configuration set (two heat exchangers), as can be seen in Table 1. Pipes, pumps and heat exchangers external surfaces were considered adiabatic.

Due to the low temperature of water during testing the thrust pad average temperature reached only 77.3 °C, with a highest value of 78.5 °C in pad no. 10.

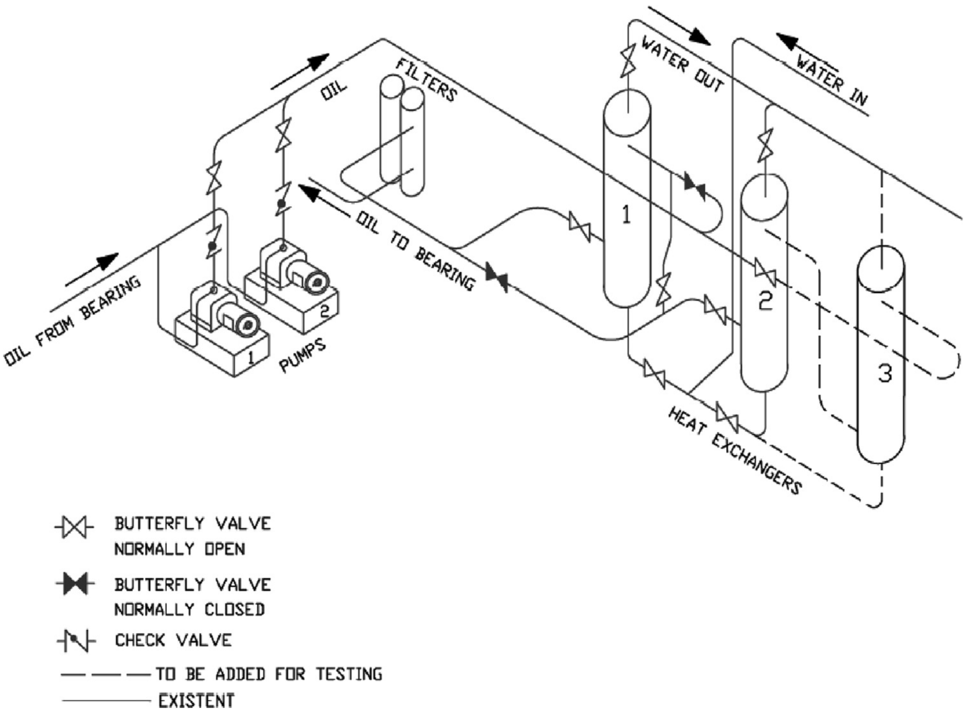

Fig. 1. Combined bearing oil refrigeration system.

Table 1
Main operational systems present in the experiment.

Parameter	Value
Generator output	285 MW
Thrust bearing pads average temperature	$\overline{T}_s = 77.3\ ^\circ\mathrm{C}$
Oil temperature at bearing inlet	$T_{o5} = 37.0\ ^\circ\mathrm{C}$
Oil temperature at bearing outlet	$T_{o1} = 45.0\ ^\circ\mathrm{C}$
Water temperature at heat exchangers inlet	$T_{wl} = 23.4\ ^\circ\mathrm{C}$
Cooling water flux per heat exchanger	$\dot{V}_w = 0.0426\ \mathrm{m^3\ s^{-1}}$
Pump differential pressure	$\Delta p = 192.6\ \mathrm{kPa}$

The relations to calculate the required power and the volumetric flow rate by the oil pump for a third heat exchanger were obtained from the data included in the product description published by the pump manufacturer (IMO) and properties of the Mobil DTE Heavy lubricant provided by Mobil Oil.

(a) Oil volumetric flow rate versus pump differential pressure:

$$\dot{V}_o = -1.707 \times 10^{-12}\Delta p^3 + 4417 \times 10^{-9}\Delta p^2 - 4586 \times 10^{-6}\Delta p + 0.03473, \tag{1}$$

where $\dot{V}_o$ ($\mathrm{m^3\ s^{-1}}$) is the oil volumetric flow rate and Δp (kPa) is the pump differential pressure. This equation is valid only if the oil temperature at the pump inlet lies between 40 °C and 48 °C and leads to a maximum error of 0.7% at a differential pressure of 1200 kPa, being smaller at lower differential pressures.

(b) Oil Mobil DTE Heavy properties:

$$\rho_o = -0.600T_o + 888.5, \tag{2}$$

$$\nu_o = 8.8 \times 10^{-8}T_o^2 - 1.124 \times 10^{-5}T_o + 3.937 \times 10^{-4}, \tag{3}$$

$$c_o = 0.00347T_o + 1.801, \tag{4}$$

where ρ_o ($\mathrm{kg\ m^{-3}}$) is the density, ν_o ($\mathrm{m^2\ s^{-1}}$) is the kinematic viscosity, c_o ($\mathrm{kJ\ ^\circ C^{-1}\ kg^{-1}}$) is the specific heat capacity and T_o (°C) is the oil temperature. These equations are valid for 40–55 °C oil temperature range.

(c) Power required to drive the oil pump:

$$\dot{W} = 0.03475\Delta p - 3.377 \times 10^8\nu_o^2 + 9.572 \times 10^4\nu_o + 1.921, \tag{5}$$

where $\dot{W}$ (kW) is the power required to drive the oil pump. This equation is valid only if the oil temperature at the pump inlet lies between 40 °C and 48 °C and leads to a maximum error of 1.0%.

Fig. 2 shows the oil cooling system of São Simão Unit no. 1 in its original configuration with two heat exchangers in series in oil and parallel in water, and the presented values to Unit 1 in operation at nominal output (see Table 1). In that configuration the oil volumetric flow rate can be calculated by Eq. (1) with $\Delta p = 192.6$ kPa, $V_o = 0.0340\ \mathrm{m^3\ s^{-1}}$.

The heat transferred in the combined guide and thrust bearing can be calculated by Eq. (6) where the oil properties are determined for the average temperature of 41 °C:

$$\begin{aligned}\dot{Q}_b &= \rho_o\dot{V}_o c_o \Delta T_o \\ &= 863.9 \times 0.034 \times 1.942 \times (45.0 - 37.0) \\ &= 456.3\ \mathrm{kW}.\end{aligned} \tag{6}$$

With the oil inlet at 45 °C at the pump, the required power can be calculated using Eq. (5), leading to $\dot{W} = 13.47$ kW.

In the pumping process, heat transferred to the oil can be calculated by applying the first law of thermodynamics to the pump's control volume

$$\dot{Q}_p = \dot{W} - \dot{V}_o\Delta p = 13.47 - 0.0340 \times 192.6 = 6.92\ \mathrm{kW}, \tag{7}$$

where $\dot{W}$ (kW) is calculated by Eq. (5) and V_o ($\mathrm{m^3\ s^{-1}}$) by Eq. (1), with Δp in (kPa).

The oil temperature at the inlet of the first heat exchanger may be derived by applying

$$\begin{aligned}T_{o2} &= T_{o1} + \frac{\dot{Q}_p}{\rho_o\dot{V}_o c_o} \\ &= 45.0 + \frac{6.92}{861.5 \times 0.034 \times 1.956} = 45.1\ ^\circ\mathrm{C}.\end{aligned} \tag{8}$$

With the oil properties calculated by Eqs. (2)–(4).

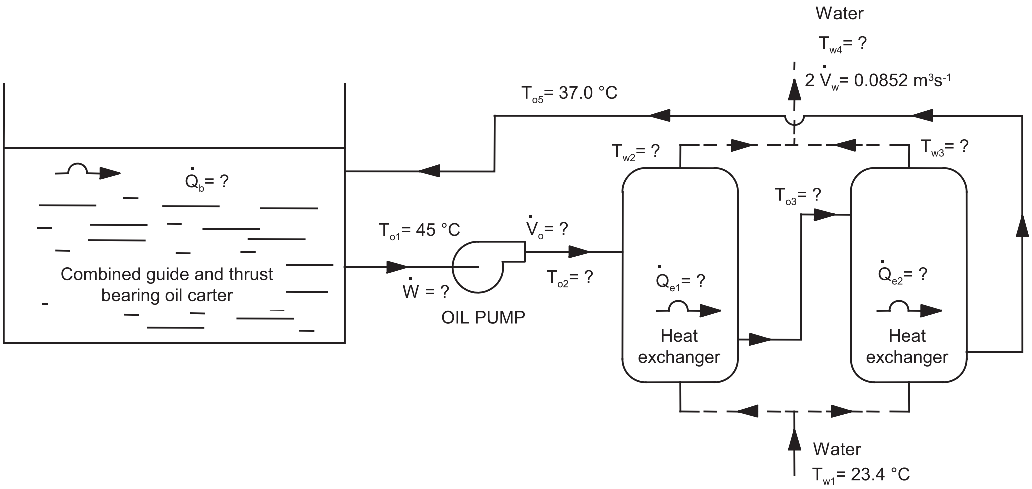

Fig. 2. Oil cooling system in its original configuration.

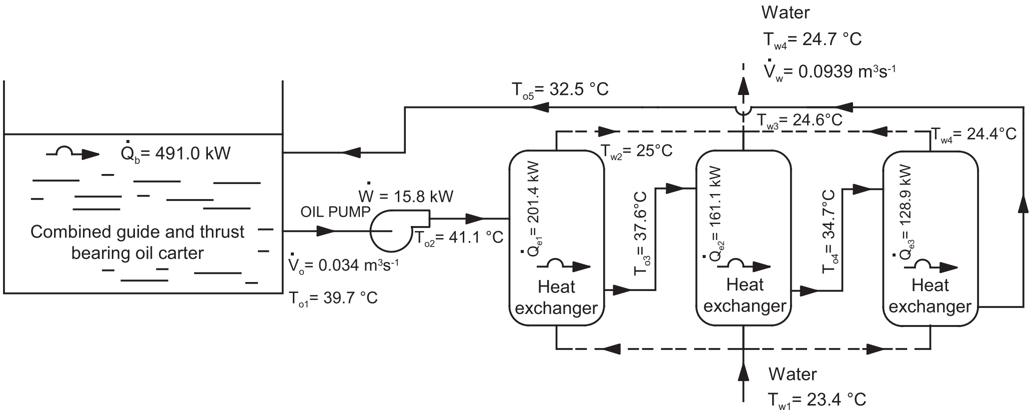

Fig. 3. New configuration for the oil cooling system.

In the heat exchangers the oil transfers a heat flow $\dot{Q}_o$ to the water, which is calculated as follows:

$$\dot{Q}_o = \dot{Q}_b + \dot{Q}_p = 463.2\ \text{kW}. \tag{9}$$

Based on classical heat exchangers theory [2,3], and using the above data, the following results are presented:

- $T_{o3} = 40.6\ ^\circ\text{C}$, oil temperature at the second heat exchanger inlet.
- $LMTD_1 = 18.7\ ^\circ\text{C}$, logarithmic mean temperature difference in the 1st heat exchanger.
- $LMTD_2 = 14.8\ ^\circ\text{C}$, logarithmic mean temperature difference in the 2nd heat exchanger.
- $\dot{Q}_{e1} = 258.8$ kW, heat flow in the 1st heat exchanger.
- $\dot{Q}_{e2} = 204.8$ kW, heat flow in the 2nd heat exchanger.
- $U_e^* = UA = 13.85\ \text{kW}\ ^\circ\text{C}^{-1}$, modified overall coefficient of heat transfer for each heat exchanger.

The insertion of the 3rd heat exchanger was evaluated through the same methodology mentioned above, under the following premises:

- The additional loss in the oil derived from the 3rd heat exchanger insertion is 47.3 kPa, as measured in a specific test [4], so the differential pressure in the pump is $\Delta p = 192.6 + 47.3 = 239.9$ kPa.
- The heat flow transferred to the oil in the pump will be computed from the new values of the inlet pump oil temperature (T_{o1}) and differential pressure (Δp).
- The water volume flow in each heat exchanger will be reduced to $0.0313\ \text{m}^3\ \text{s}^{-1}$ (CEMIG's request), leading to $0.0939\ \text{m}^3\ \text{s}^{-1}$ for the three heat exchangers.
- The modified overall coefficient of heat transfer will be computed for the new water volume flow.

- The heat exchangers inlet water temperature will remain the same, i.e., 23.4 °C.
- The difference between the average thrust pad temperature and the oil temperature in the bearing inlet will remain the same, i.e., $\Delta T_{so} = \overline{T}_s - T_{o4} = 77.3 - 37.0 = 40.3$ °C.
- The expected thrust pads average temperature is

$$\overline{T}_s = T_{o5} + \Delta T_{so}. \tag{10}$$

- The heat flow dissipated in combined thrust bearing will vary with the square root of the oil coefficient of viscosity in the oil-film [5].

2. Results

Fig. 3 shows the proposed configuration for the oil cooling system with the insertion of the 3rd heat exchanger, in series with the two existents.

Table 2 compares the predicted values with experimental ones obtained by CEMIG for Unit no. 1 in steady-state operation at nominal output.

A new computation, which is done for the real water temperature at the heat exchangers inlet, 23.8 °C, and pressure difference, 224.0 kPa, gives the values presented in Table 3.

The small difference between the predicted and measured temperature values of both the oil and the pads demonstrates the model reliability.

Another computation was done to consider the highest inlet water temperature ever registered at the São Simão Power Plant, the results of which are shown in Table 4.

Table 2
Predicted and measured values comparison.

Parameter	Values	
	Predicted	Measured
T_{w1}: Water temperature at heat exchangers inlet, °C	23.4	23.8
$\dot{V}_w$: Water volume flow in each heat exchanger, m^3 s^{-1}	0.0313	0.0313
Δp: Pressure difference in the pump, kPa	239.9	224.0
T_{o5}: Oil temperature at combined bearing inlet, °C	32.5	33.5
T_{o2}: Oil temperature at pump outlet, °C	41.1	42.0
$\overline{T}_s$: Average temperature of thrust pads, °C	72.8	74.8

Table 3
Calculated and measured values—a new computation for the same input, $T_{w1} = 23.8$ °C and $\Delta p_o = 2.240$ bar.

Parameter	Values	
	Predicted	Measured
T_{w1}: Water temperature at heat exchangers inlet, °C	23.8	23.8
$\dot{V}_w$: Water volume flow in each heat exchanger, m^3 s^{-1}	0.0313	0.0313
Δp: Pressure difference in the pump, kPa	224.0	224.0
T_{o5}: Oil temperature at combined bearing inlet, °C	32.9	33.5
T_{o2}: Oil temperature at pump outlet, °C	41.5	42.0
$\overline{T}_s$: Average temperature of thrust pads, °C	73.2	74.8

Table 4
Predicted values with $T_{w1} = 30.8$ °C.

Parameter	Predicted
T_{w1}: Water temperature at heat exchangers inlet, °C	30.8
$\dot{V}_w$: Water volume flow in each heat exchanger, m^3 s^{-1}	0.0313
Δp: Pressure difference in the pump, kPa	224.0
T_{o5}: Oil temperature at combined bearing inlet, °C	39.3
T_{o2}: Oil temperature at pump outlet, °C	47.3
$\overline{T}_s$: Average temperature of thrust pads, °C	79.6

The average pad temperature predicted was 79.6 °C, which is sufficiently lower than 84 °C, the temperature at which the failures occurred. In all the cases, the simulated pressure difference and the power required for the oil pump lie within the technical limits recommended by the manufacturer for the pump and driving motor.

3. Conclusions

The insertion of an on hand heat exchanger in a new configuration was an excellent solution because it generated a good performance in a short amount of time with very low cost. The test performed using three heat exchangers rather than two confirmed the models predicted values for the oil and thrust pad temperatures with a satisfactory precision and allowed the Unit no. 1 to operate at nominal output with no restrictions in a timely manner.

CEMIG extended the alternative solution to the other machines, where the original shell-and-tubes heat exchangers were changed to stainless steel plate ones. In spite of changes performed in the oil cooling system, the oil pump and its driving motor lied inside the limit values prescribed by its manufacturer. The total direct cost was US $600,000.00 (six hundred thousand dollars) for the six machines, with a significant savings of US $1,800,000.00 (one million and eight hundred thousand dollars) when compared with the solution proposed by the bearing manufacturer.

Furthermore, the estimated time for the bearing manufacturer's proposal implementation was a minimum of approximately forty-five days while CEMIG's alternative was implemented in fifteen days, thus reducing indirect costs considerably in applying the alternative solution.

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